Correction - September Issue

We apologize for the mistake in H.H. Jeffcott's equations of motion for his idealized rotor model which were presented in the September Orbit. Many people (but not your editor) caught the incorrect use of the cosine function in the equation for motion in the y direction, when it should have been the sine function:

$$m\ddot{x} + b\dot{x} + cx = ma\omega^2 \cos\omega t$$

$$m\ddot{y} + b\dot{y} + cy = ma\omega^2 \sin\omega t$$

While these equations corrected a major misconception and set rotor dynamics back on the proper course, they fail to take into account the tangential action of the fluid in the bearings or seals. This action was empirically identified as early as the 1920's, but was not formalized into the equations of synchronous motion until the 1980's by Bently Rotor Dynamics Research Corporation:

$$M\ddot{x} + D_s\dot{x} + (K + K_B)x + D(\dot{x} + \lambda\Omega y) = m\,r\Omega^2 \cos(\Omega t + \phi)$$

$$M\ddot{y} + D_s\dot{y} + (K + K_B)y + D(\dot{y} + \lambda\Omega x) = m\,r\Omega^2 \sin(\Omega t + \phi)$$

where M, D_S and K are the rotor system modal mass, damping, and stiffness, respectively; D and K_B are the fluid radial damping and stiffness, respectively; Ω is rotative speed; λ is the Fluid Circumferential Average Velocity Ratio (the "dashpot" representing the system damping is not stationary, but rotates at the angular velocity $\lambda\Omega$); m, r, and ϕ are the modal mass, radius, and phase of the unbalance, respectively. The $D\lambda\Omega$ terms represent the cross-coupled stiffness due to the action of the fluid, the effect of the fluid pushing on the shaft, which is highly dependent on the rotative speed. The $D\dot{x}$ and $D\dot{y}$ terms are the traditional viscous damping terms, the effect of the shaft pushing on the fluid. A more complete description of these equations and their effects can be found in the December 1994 Orbit article by Agnes Muszynska and Don Bently, "Fluid dynamic force model for rotors with seals or lightly-loaded bearings."